NATURAL CONVECTION HEAT TRANSFER FROM A HORIZONTAL DISK IN A CYLINDRICAL ENCLOSURE

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THESIS

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December 1971

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Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
December 1971

ABSTRACT

This thesis studied natural convection in water from a horizontal mirror finished disk, heated in the center and surrounded by a cylindrical enclosure. The test section was designed to insure a one dimensional heat flux. Measurements were taken with power inputs varying from 25 to 5 watts.

Fluid depth had a definite effect on the heat transfer. An apparent maximum value existed at a ratio of enclosure radius to fluid depth of one. Comparison of the data with existing correlations however, was poor which led to the questioning of the validity of the assumed one dimensional heat flux. Use of a finite element computer program demonstrated that two dimensional effects were important. Subsequent modification of the heat transfer coefficient to account for this variation gave a correlation more in agreement with those existing in the literature.

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NOMENCLATURE

Symbol	Description	Units
A	Appropriate area	Ft^2
d	Heated surface diameter	inches or ft
F	Form factor of radiation heat loss	
g	Standard acceleration of gravity	ft/sec^2
h	Convection heat transfer coefficient	$Btu/Hr\text{-}Ft^2\text{-}°F$
h'	Modified convection heat transfer coefficient	$Btu/Hr\text{-}Ft^2\text{-}°F$
k	Thermal conductivity	$Btu/Hr\text{-}Ft\text{-}°F$
L	Characteristic Dimension	Ft.
P	Perimeter of Fin	Ft.
Q	Heat rate	Btu/Hr or watts
Q_r	Radiation heat loss	Btu/Hr
Q/A	Heat Flux	$Btu/Hr\text{-}Ft^2$
T	Thermocouple Temperature	°F
T_s	Test surface temperature	°F or °R
T_b	Fluid bulk temperature	°F
T_f	Film temperature $(T_s + T_b)/2$	°F
T_w	Test fluid temperature	°R
$\Delta T/\Delta x$	Test section axial temperature gradient	°F/Ft
α	Fluid thermal diffusivity	Ft^2/sec
β	Fluid coefficient of thermal expansion	$°F^{-1}$
ε	Radiation emissivity	

σ	Stefan-Boltzman Constant	$Btu/Hr\text{-}Ft^2\text{-}{}^\circ R^4$
ν	Fluid kinematic viscosity	Ft^2/sec

Dimensionless Numbers

Gr_d	Grashof number based on heated test section	$g\beta d^3 \Delta T/\nu^2$
Gr_L	Grashof number based on some characteristic length	$g\beta L^3 \Delta T/\nu^2$
Nu_d	Nusselt number based on heated test section	$h\ d/k$
$(Nu_d)'$	Modified Nusselt number based on heated test section	$h'\ d/k$
Nu_L	Nusselt number based on some length	$h\ L/k$
Pr	Fluid Prandtl number	ν/α
Ra_d	Rayleigh number	Gr Pr

ACKNOWLEDGEMENT

This author wishes to express his sincere appreciation to his wife Lynn, whose understanding and good humor made the difficult times endurable. He also wishes to thank Professor P. J. Marto for his patience and guidance in overcoming the difficult times.

A special word of sincere thanks to Professor P. F. Pucci for the use of his conduction finite element computer program, without which this project may never have been completed.

I. INTRODUCTION

In recent years there has been a proliferation of investigations on natural convection. Much of this research has been done with vertical surfaces, since this geometry is easier to analyze. The study of natural convection from horizontal surfaces has been less emphasized, although this geometry is quite common during pool boiling experiments.

Moulson [1] obtained natural convection data from a horizontal disk in a cylindrical enclosure while studying pool boiling in liquid Nitrogen. Data was obtained using both a mirror finish surface and a surface containing artificial cavities. Moulson's investigation indicated that there was an increase in the heat transfer coefficient for the surface containing artificial cavities. However, as reported by Moulson [1], there was a large uncertainty in the results obtained in the natural convection region of his study.

An attempt to find an applicable theoretical study for a geometry similar to Moulson's, led to consideration of papers by Lighthill [2], Brentwich [3], Ostrach [4], Torrance, et al [5], and others. However, with the exception of Torrance [5], the geometrical models did not sufficiently approximate the experimental set up used by Moulson.

Torrance, et al [5] presented a numerical study of natural convection of air in an enclosure similar to the one used by Moulson, i.e., a circular plate with a heated section in the center, surrounded by a cylindrical enclosure. In the numerical solution, the authors used the parameters of the ratio of the radius of the enclosure to fluid depth (aspect ratio), and the ratio of the heated radius to fluid depth. These two ratios were then used in the boundary conditions for the numerical solution of the governing equations. Torrance reported results, however, for only a single geometry with an aspect ratio of one and a heated radius to fluid depth ratio of 0.1.

Subsequently, this investigator reviewed existing empirical correlations of the form

$$Nu=C(GrxPr)^n \tag{1}$$

where Nu is the Nusselt number and Gr is the Grashof number based on some characteristic length. Fishenden and Saunders [6] obtained data for natural convection in air from a large horizontal flat plate which was represented by

$$Nu_L=hL/k=0.54(Gr_L Pr)^{1/4}$$

for $(GrPr) < 2x10^7$

and

$$Nu_L=hL/k=0.14(Gr_L Pr)^{1/3}$$

for $(GrPr) > 2x10^7$.

Heled, et al [7] obtained natural convection data for various hydrocarbons from a circular horizontal surface in a cylindrical chamber. They reported that values of the coefficient in equation (1) ranged from 0.14 to 0.32.

The objective of this study was to redesign the test apparatus used by Moulson [1] in order to obtain reproducible natural convection data from a mirror finish surface using water as the test fluid, and compare these results with those obtained by other investigators. Such data could then be used for further investigation to see what effect perturbations of the surface such as artificial cavities have on natural convection heat transfer.

II. EXPERIMENTAL APPARATUS

The basic system as shown in Fig. 1 was designed as an improvement over that used by Moulson [1]. The test section was redesigned so that the test surface could be placed in the center of the dewar. The heater and thermocouple wires were led out the bottom of the test section through the connecting tube, thus improving the symmetry of the test section in the dewar. The test section was manufactured of stainless steel to give a large axial temperature gradient. In Moulson's study a copper section was used to obtain an axial temperature gradient. The high thermal conductivity of the copper resulted in a small temperature gradient and thus was a significant factor in the high uncertainty of Moulson's data for natural convection.

A double glass dewar, silvered on both sides was used to hold the fluid. A dewar was used in order to provide the capability for testing with a variety of fluids, from liquid Nitrogen to more conventional types.

A. TEST SECTION

The test section as shown in Fig. 2 was fabricated from a single piece of 304 stainless steel. The top diameter was three inches with a 1.5 inch section one inch in diameter extending down. The flat disk on top was machined to 0.030 inches thick adjacent to the one inch section in order to minimize radial conduction losses. Four nominal

1/8 inch holes were drilled to the centerline of the cylinder, 90° apart, for thermocouple installation. Centerline distances from the test surface are given in Fig. 2. The distances were determined by placing the test section on gage blocks, inserting the thermocouple plugs described in section B and measuring the distance with a scale readable to 0.0005 inch. The bottom of the holes were end milled to insure a good contact surface for the thermocouples. A hollowed out copper block was silver soldered to the base of the one inch cylinder to distribute the heat flux uniformly across the base of the test section. A 250 watt, 115 volt resistance heater was inserted in the copper block as the heat source, and connected to a D.C. power supply.

B. TEMPERATURE MEASUREMENT SYSTEM

Since one dimensional heat flow was a basic assumption in this study, the precision of the temperature measurements was considered important. Heat flow through the test surface was determined by four thermocouples placed along the centerline of the 1 1/2 inch test section as indicated in section A.

The thermocouples used were fiberglass insulated, three mil copper-constantan. This size was chosen to help reduce the uncertainty in the junction location. The junctions were formed using resistance welding.

The copper-constantan wires were connected to one side of a D.C. power supply and a copper block was

connected to the other side. The junction was formed by contacting the copper block with the thermocouple wires. Nitrogen gas was used to provide an inert atmosphere to minimize oxidation and provide a stronger junction.

The thermocouples were placed in 304 stainless steel plugs shown in Fig. 3. The junction was soft soldered in place and epoxied at the other end to prevent loosening after installation. The threaded section was used to insure good contact by the thermocouple with the test section.

The location of the thermocouple junction in the 0.010 inch hole in the plug could not be determined accurately. The maximum uncertainty would be $\pm$ 0.005 inches for the hole. The most probable uncertainty would be less than this. For this study the thermocouple junction was assumed to be in the center of the 0.010 inch hole and the location known to $\pm$ 0.003 inch.

Copper-constantan extension wire was used from the air side of the vacuum connection to a Hewlett-Packard 2010C Data Acquisition System. The ice junction was also made of extension wire. Thermocouples were calibrated using a constant temperature oil bath. Appendix A outlines the procedure in detail.

Fluid temperature was measured using a thermocouple mounted in a glass tube and capable of being positioned axially and radially.

C. VACUUM SYSTEM

The test section can was evacuated to reduce heat loss due to convection in the can and tube.

A Welsh Duo-Seal Vacuum Pump was used as a roughing pump. An air cooled diffusion pump using Dow-Corning 705 Silicone Oil was used to maintain the vacuum. The diffusion pump was protected by a high temperature cut-out relay that functioned at 110°C. The vacuum system was connected to the test stand by means of rubber tubing in order to minimize vibrations to the test stand from the vacuum system roughing pump.

A minimum vacuum of 5×10^{-5} mm Hg at the base of the tube was maintained during all tests. Van Atta [8] and Barrington [9] indicate that at vacuums of the order of 10^{-3} to 10^{-4} mm Hg the radiation effects are of the same order of magnitude as convection effects.

III. EXPERIMENTAL PROCEDURE

A. TEST SURFACE PREPARATION

In order to minimize any surface effects on the heat transfer, the test surface was finished to a mirror finish as follows. The test surface was hand sanded successively on 0, 2/0, and 3/0 emery paper. The surface was washed with soap and water between each grade of paper to remove any grit.

After dry polishing, the test surface was wet polished on three metallurgical wheels covered with velvet and impregnated with 6, 3, and 1 micron diamond dust and lubricated with methanol. The surface was washed with soap and water and rinsed with methanol between each wheel.

B. TEST SECTION INSTALLATION

The test section was installed in the can after the thermocouple plugs and heater leads had been put in place. A teflon seal was used between the can and the test section. The section was secured to the can by eight 3/16 inch stainless steel studs which were torqued to a maximum of 35 inch pounds.

C. TEST SURFACE WETTING

The test surface was initially cleaned with acetone and alcohol prior to filling the dewar with water. However, it was found during run one that the heated surface was not wet. Subsequent data was taken after the following

cleaning procedure was used to promote wetting of the test surface by the water. The surface was cleaned two times with trichlorethylene, then rinsed three times with 190 proof ethyl alcohol and then rinsed five times with distilled water. Then the surface was soaked with a solution consisting of eight parts water, two parts 190 proof ethyl alcohol, two parts 50% NaOH, and one half part 30% H_2O_2. This solution was allowed to cover the test surface for ten minutes. The surface was then rinsed six times with distilled water and left wet. The dewar was installed and filled immediately with water.

D. FLUID PREPARATION

The fluid used in all tests was distilled water. The dewar was filled with water to a depth of approximately eight inches. The heater was then set to 25 watts. After there were noticible convection currents from the test surface a submersible heater was placed in the dewar and the water brought to a boil. The water was allowed to boil freely from the test surface and the heater for 30 minutes to eliminate dissolved air. Then the heater was removed and the system allowed to come to equilibrium.

E. TESTING PROCEDURES

Four data sets were obtained using water, and one set of readings was taken to determine heat loss. Two runs were made with varying power inputs with the water depth held constant at 7.35 inches. One run was made with a

varying power input at a water depth of 1.76 inches. The depth of 7.35 inches was used since it effectively filled the dewar. The depth of 1.76 inches was used to give a ratio of enclosure radius to depth of one, for comparison with the numerical solution of Torrance, et al [5]. Power settings were 25.0, 20.0, 15.0, 10.6, and 5.0 watts. Data points were taken starting with the highest power setting and decreasing to the next setting. This was done so that each data point was at a successively lower fluid temperature, and consequently the solubility of air in the fluid would be increasing. It was hoped that this procedure would help eliminate bubble formation at the fluid test surface.

One run with water was made with a constant power input of 5.0 watts and a varying water depth of 7.35, 5.50, 3.60, and 1.76 inches to see if a determination could be made of the effect of fluid depth on natural convection heat transfer.

1. Temperature Determination

The temperatures in the test section were obtained by taking a series of five reading on the Data Acquisition System for each thermocouple. These millivolt readings were then averaged and the corresponding temperature read from the Thermocouple Calibration Curves.

The bulk temperature of the fluid in the dewar was determined using the thermocouple probe shown in Fig. 1. A series of five readings were taken at each of six locations. Three readings were taken at approximately 1/2 inch above

the test surface, and three readings at approximately 2/3 of the depth of the fluid. The readings at each location were averaged and then these six averages averaged and a temperature read from the calibration curve.

2. Steady State Determinations

Since stainless steel was used to insure a strong axial gradient, the time to reach equilibrium was quite long. On the first run, the time allowed between data points was approximately 24 hours. On subsequent runs however, readings were taken initially every two to four hours and the rate of change of the temperature with time recorded. When the rate decreased to less then one degree per hour, readings were taken hourly until the change was less than 0.2 degrees per hour. When the readings changed less than 0.2 degrees per hour the system was considered to be at steady state.

3. Heat Flux

The heat flux through the test surface was determined using a one dimensional form of the Fourier Conduction equation,

$$Q/A = -k\ \Delta T/\Delta x \qquad (2)$$

where k, the thermal conductivity in Btu/Hr-Ft.°F was assumed constant, and $\Delta T/\Delta x$ was determined from the slope of the linear least squares fit of the thermocouple readings and locations.

The heat transfer coefficient was determined using the formulation,

$$Q/A = h(T_s - T_b) \qquad (3)$$

T_s, the surface temperature, was determined by extrapolating the temperature versus distance curve to the surface location. This value was assumed to be constant over the test section.

4. <u>Heat Loss</u>

In addition to the runs made with water, a run was made to try and determine the amount of heat loss by radial conduction in the test section. A cardboard cylinder was cut to the inside dimensions of the dewar and was 14 inches long. It was filled with three layers of Johns-Mansville MIN-K insulation and loose asbestos fiber. Water was placed in the dewar up to the level of the test surface and the insulating block was inserted. Readings were taken for power inputs of 0.03, 0.86, 2.2, 3.4, 4.9, and 6.7 watts. Figure 4 is a plot of the results, where the average temperature is defined as the arithmetic average of the four centerline thermocouple readings.

IV. RESULTS

A. REPRODUCIBILITY

Moulson [1] reported that, for natural convection data with liquid Nitrogen from a mirror finished test surface, there was a wide discrepancy in the results for the same inputs. Figure 5 is a plot of the heat transfer coefficient, h, defined in Eq.(2)versus (T_s-T_b) for the two runs made with a water depth of 7.35 inches. The agreement was good despite the fact that in run one the surface was covered with air bubbles at all power inputs, and run two was done after the promotion of wetting, with no bubbles. The maximum uncertainty in h was calculated to be less than 6%.

B. EFFECT OF FLUID DEPTH

Figure 6 is a plot of the heat transfer coefficient versus fluid depth for a constant input of five watts. The lowest point represents the limiting case where the heat transfer would be by pure conduction. This point was calculated as defined in Gebhart [10].

The variation in the heat transfer coefficient with depth was considered plausible. In the region shown in Fig. 6 by a dotted line, an increase in depth would cause the onset of natural convection currents, with a resulting increase in the heat transfer coefficient. At some point the convection currents would form a regular flow cell, and

a maximum natural convection heat transfer coefficient would be reached. In this study, this occurred at a fluid depth to diameter ratio of approximately one. Any further increase in depth would cause interference between decending cool fluid and rising warm fluid, disturbing the flow and reducing the heat transfer coefficient. This trend is in agreement with data obtained by Bayley, F. J., et al [11] in a study with liquid metals.

C. COMPARISON WITH OTHER CORRELATIONS

A plot of the Nusselt number versus Rayleigh number for the three runs made with varying power inputs is shown in Fig. 7. The straight line is a least squares fit for the data points of all three runs. The resultant correlation is given in the form

$$Nu_d = 2.1(Ra_d)^{0.20} \quad (4)$$

where Nu_d is the Nusselt number based on the experimental heat transfer coefficient calculated using Eq. (3), and the subscript refers to the characteristic dimension which in this case is the diameter of the heated surface. Ra_d is the Rayleigh number, or Grashof number Prandtl number product, also using the heated test surface diameter as the characteristic dimension. This correlation was outside the range of values reported by Heled, et al,[7].

A comparison of the results of Torrance, et al,[5] with this investigation is shown in Fig. 8. It should be noted that the results given by Torrance were for a single geometry,

aspect ratio equal to one, and the ratio of the heated radius to fluid depth of 0.1. In this work, run one and two had an aspect ratio of 0.24 and a heated radius to fluid depth ratio of 0.07. Run three had an aspect ratio of one and a heated radius to fluid depth ratio of 0.28.

The results of the correlation and the plot in Fig. 8 indicated that the value of the heat transfer coefficient was too high, suggesting the method of obtaining the heat flux, Q/A, used in determining the heat transfer coefficient was invalid. A comparison was therefore made of the power out computed from Eq. (2) and the net power out based on the power in and the heat loss given in Fig. 4. Table I gives the results of this comparison.

TABLE I

Power Loss Summary

Q in (Watts)	T avg. (°F)	Q Loss (Fig. 4) (Watts)	$Q_{in}-Q_{loss}$ (Watts)	$Q=-kA\Delta T/\Delta x$ (Watts)
Run 1 - Water Depth 7.35 Inches				
25.0	286.9	~ 9.0	~ 16.0	22.3
20.0	260.8	6.9	13.1	18.1
15.0	220.9	5.0	10.0	13.5
10.6	186.4	3.7	6.9	9.8
5.0	137.5	2.0	3.0	4.7
Run 2 - Water Depth 7.35 Inches				
25.0	187.6	~ 9.0	~ 16.0	22.4
20.0	259.6	7.1	12.9	18.1
15.0	225.2	5.2	9.8	13.5
10.6	188.9	3.8	6.8	10.0
5.0	139.4	2.1	2.9	4.6
Run 3 - Water Depth 1.76 Inches				
25.0	296.2	~ 9.0	~ 16.0	22.3
20.0	266.6	7.5	12.5	18.5
15.0	232.1	5.5	9.5	13.4
10.6	195.7	4.0	6.6	9.9
5.0	144.7	2.3	2.7	4.9

It is apparent from Table I that there was a significant discrepancy between the calculated experimental values of the power out and those based on the heat loss experiment. There appeared to be two possibilities for this discrepancy, (1) the heat loss determination was not valid, and/or (2) the assumed uniform surface temperature distribution over the test section was in error.

Use was made of a conduction finite element computer program, developed by Professor Paul F. Pucci, Department of Mechanical Engineering at the Naval Postgraduate School, to generate the temperature distribution in the test section. Figure 9 shows the model of the test surface used, with the assumed boundary conditions. Since the program required that the boundary conditions be defined over the entire surface, and these were in effect unknown, the program was not used to get quantitative results, but rather to see if the assumed uniform temperature distribution was valid. Heat inputs of 5 and 25 watts were chosen as the lower and upper bounds. The heat flux into the test section was assumed constant across the base. Radiation losses were neglected. (Appendix C indicates that these losses were less than 5%). However, the heat flux into the section was based on that determined from Eq. (2). This value was approximately 90-95% of the heat input. The convection heat transfer coefficient, h, along the upper surface was assumed constant across the 0.5 inch radius test section and was then assumed to decrease linearly to a value of 80 Btu/Hr ft^2 °F at the outer edge of the surface. The

value of h on the vertical surface was assumed constant and equal to 80 Btu/Hr-Ft^2-°F. The fin effect of the studs was taken into consideration in calculating the equivalent h = 52 Btu/Hr-Ft^2-°F for the underside of the surface. Appendix D gives the methods of computation of the above values for h. Computer runs were made varying the value of h along the upper surface as a fraction of the experimental value, in order to match the surface temperature at the centerline with the experimental value. Figures 10 and 11 show the resultant surface temperature distribution. The same procedure was used for one run of the heat loss experiment at 4.9 watts. The top surface was insulated and an arbitrary fluid temperature assumed. The results are shown in Fig. 12.

The temperature difference shown in Fig. 12 between the surface temperature at the outer edge of the test section indicates that the heat loss measurements were not valid when compared to the runs with water. Insulating the top surface forces a large amount of heat in a radial direction and also raises the overall temperature so that radiation could become important.

It is also apparent that the assumption of a uniform surface temperature distribution across the test section is not valid. There is sufficient radial conduction to cause a non uniform temperature distribution and consequently a two dimensional problem.

D. MODIFIED HEAT TRANSFER COEFFICIENT

In an attempt to verify the preceding results, a modified heat transfer coefficient, h' was used. The modified values were obtained by using the values from the 5 watt and the 25 watt power input cases for run one, reduced by 35% and 20% respectively, and drawing a straight line between them (Fig. 13). Subsequent points were then taken from this curve for the other points and runs. Table II gives an example of the values obtained for run one.

TABLE II

Run 1

Input (Watts)	h Experimental ($Btu/Hr\text{-}Ft^2\text{-}^\circ F$)	h' (Modified) ($Btu/Hr\text{-}Ft^2\text{-}^\circ F$)
5	191.8	125.0
10.6	263.9	194.0
15.0	282.2	211.0
20.0	320.0	247.0
25.0	376.8	301.0

The resulting modified Nusselt number $(Nu_d)'$ versus Grashof number is plotted again in comparison with that of Torrance in Fig. 14. The agreement has been improved. Figure 15 is a plot of the modified Nusselt number versus Rayleigh number. The resultant correlation obtained was

$$(Nu_d)' = 0.31(Ra_d)^{0.30} \qquad (5)$$

which agrees well with the data of previous investigations.

These results further indicate that the assumption of a uniform surface temperature is not valid for the test section used.

V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

1. With the test section used in this study, the assumption of one dimensional heat flux is not valid.

2. The depth of the fluid above the test section has a definite effect on the heat transfer. There is an apparent maximum value of the heat transfer coefficient which is determined by the ratio of enclosure radius to fluid depth.

3. Using a heat transfer coefficient modified to reflect the effect of radial conduction in the test surface it was possible to obtain a correlation of the form $Nu_d = 0.31 (Ra_d)^{0.30}$ which is in agreement with earlier reported results.

B. RECOMMENDATIONS

The following recommendations are made for future studies:

1. The test surface be redesigned to eliminate radial conduction, thus obtaining a uniform temperature distribution across the test section. The use of a method of inserting a glass insulation ring could be studied if the capability of using cryogenic fluids is to be maintained or, if only conventional fluids are to be studied, the vacuum system could be eliminated and the test section isolated by a ring of conventional insulation material.

In addition foil insulation could be used to further insure that radiation heat losses are eliminated.

2. Obtain analytical results using the numerical program, developed by Professor Pucci, in conjunction with experimental data to deduce the measured value of the heat transfer coefficient.

3. Obtain analytical results using the numerical program of Torrance with different fluid depths, to see if his results can be generalized.

APPENDIX A

THERMOCOUPLE CALIBRATION

Since a high degree of accuracy was desired, it was felt that the thermocouples should be calibrated over the entire range of expected temperatures, from 70°F to 400°F.

To accomplish the calibration, a Rosemount Calibration System was used. This system used a variable temperature oil bath with a listed accuracy of ± 0.002°C. A constant temperature ice bath with a listed accuracy of ± 0.002°F was used for the ice junction. A Platinum Resistance Thermometer connected to a commutation bridge with a maximum error in the temperature range of 0.004 C was used as the standard.

An initial calibration was performed with the five thermocouples suspended in glass tubes in the oil bath. The calibration was not satisfactory due to convection currents in the tubes. A subsequent run was made with the thermocouples suspended in an aluminum cylinder packed with ultrafine aluminum dust. The validity of this calibration was considered suspect because of the possibility that a temperature gradient existed between the standard and the thermocouples, and the difficulty in determining when an

equilibrium condition was reached. Finally, the thermocouples were suspended directly in the oil bath and the calibration was completed.

The thermocouple readings were averaged and the resulting value in millivolts assigned as a calibration point for the temperature determined by the standard. A straight line interpolation was used between consecutive calibration points to plot a millivolt versus temperature curve that could be read to the nearest 0.1°F. Table III lists the calibration temperature and average millivolt readings used.

TABLE III

Thermocouple Calibration Summary

Standard Temp. (°F)	Average Thermocouple Reading (mv)
79.81	1.087
128.80	2.235
176.03	3.401
227.87	4.739
271.08	5.897
317.29	7.177
366.83	8.595
403.48	9.672

Using the calibration plot it was believed that the temperature could be determined to ± 0.2°F.

APPENDIX B

SAMPLE CALCULATIONS WITH ERROR ANALYSIS

A. ASSUMPTIONS

1. Based on the thermocouple calibration all thermocouples were accurate to $\pm$ 0.2°F.

2. The location of the thermocouple was known to $\pm$ 0.003 inch.

3. The thermal conductivity of the 304 stainless steel was known to 5% [12].

4. All thermocouples have the same uncertainty.

The data used in all calculations in this appendix are from point one of run one. (25.0 watts input, water depth 7.35 inches).

TABLE IV

Thermocouple Data

T. C.	Location (in)	Temp. °F
1	0.352 $\pm$ 0.003	237.0 $\pm$ 0.2
2	0.600 $\pm$ 0.003	269.5 $\pm$ 0.2
3	0.894 $\pm$ 0.003	301.2 $\pm$ 0.2
4	1.203 $\pm$ 0.003	340.0 $\pm$ 0.2
5	Bulk Fluid	158.6 $\pm$ 0.2

B. SURFACE TEMPERATURE AND HEAT FLUX DETERMINATION

Equation (1) was used to determine the heat flux Q/A. A linear least squares calculation was used to obtain the slope $\Delta T/\Delta x$ and the intercept T_s. Although the most probable error in the temperature was less than 0.2°F, this value was added and subtracted to the reading, and a maximum and minimum slope and intercept obtained. These results were used to obtain a maximum and minimum value for the heat flux. The arithmetic value was then used in subsequent calculations. Table V gives the results.

TABLE V

Sample Heat Flux and Surface Temperature

Q/A (Btu/Hr-Ft^2)		T_s (°F)
13979.0	average	195.7
14044.8	maximum	196.2
13912.9	minimum	195.3

The value of the thermal conductivity, k, used in determining Q/A was determined using the formula

$k = 9.36 + 0.0043(T - 200)$Btu/Hr-Ft-°F. $\pm$ 5%.

This formulation was determined using a linear correlation of values obtained from Ref. [12], with an uncertainty of $\pm$ 5%. T was the arithmetic average of the temperature read by the thermocouples. In this case

$k = 9.74 \pm 0.49$ Btu/Hr-Ft-°F. T_s (surface temperature) was 195.7 ± 0.5°F.

The root mean square error in Q/A was determined using the formulation

$$\frac{\delta q/A}{q/A} = \left[\left(\frac{\delta k}{k}\right)^2 + \left(\frac{\delta \Delta T}{\Delta T}\right)^2 + \left(\frac{\delta \Delta x}{\Delta x}\right)^2\right]^{1/2}$$

where

$$\delta \Delta T = \sqrt{2} \times 0.2 = 0.28°F$$

and if we consider the worst case for the thermocouple location

$$\delta \Delta x = \sqrt{2} \times 0.003 = .004 \text{ inches}$$

and

$$\frac{\delta q/A}{q/A} = \left[(0.05)^2 + \left(\frac{0.28}{103}\right)^2 + \left(\frac{.004}{1.203}\right)^2\right]^{1/2} = 0.0501$$

Thus $Q/A = 13979.0 \pm 700.7$ Btu/Hr-Ft^2.

C. NATURAL CONVECTION

To determine the natural convection parameters it was necessary to determine $T_s - T_b$.

For this case,

$$T_s - T_b = 195.7 - 158.6 \pm \left[(0.5)^2 + (0.2)^2\right]^{1/2}$$

$$= 37.1 \pm 0.54°F$$

From Eq. (3)

$$q/A = h(T_s - T_b)$$

thus

$$h = q/A/(T_s - T_b) = \frac{13979.0}{37.1} = 376.8 \text{ Btu/Hr-Ft}^2\text{-°F}$$

and the root mean square error

$$\frac{\delta h}{h} = \left[\left(\frac{\delta q/A}{q/A}\right)^2 + \left(\frac{\delta \Delta T}{\Delta T}\right)^2\right]^{1/2} = 0.052$$

thus

$$h = 376.79 \pm 19.58 \text{ Btu/Hr-Ft}^2\text{-}^\circ\text{F}.$$

It is obvious that the uncertainty in the thermal conductivity was the controlling parameter in determining the error.

The Nusselt number, $\frac{h\ d}{k}$, Grashof number $\frac{g\beta d^3 \Delta T}{\nu^2}$, and the Rayleigh number (G_r x P_r) were based on the heated test section diameter (one inch) and the fluid properties evaluated at the film temperature $\frac{T_s + T_b}{2}$. Values of the fluid properties were obtained from Kreith [13].

APPENDIX C

RADIATION LOSS

Initially, radiation losses were considered negligible. However, in view of the apparent discrepancies in heat flux, a check was made for run one of the estimated radiation heat loss. The equation,

$$Q_r = \varepsilon\sigma AF\ (T_s{}^4 - T_w{}^4)$$

was used. The geometry was assumed to be representable as two infinite parallel plates, F = 1. T_s was chosen as the average centerline temperature, T_w was the fluid temperature, and ε was chosen as 0.44 from McAdams [14].

TABLE VI

Input (Watts)	Average Temp. °R	Q_r (Watts)	% Input
25.0	746.1	1.02	4.1
20.0	720.1	0.81	4.0
15.0	681.0	0.57	3.8
10.6	646.3	0.38	3.6
5.0	598.1	0.18	3.5

This estimate of the heat loss was conservative since the surface temperature was less than the centerline and also varied along the surface. Therefore it was determined that radiation heat loss was not significant.

APPENDIX D

HEAT TRANSFER COEFFICIENT FOR COMPUTER MODEL

As shown in Fig. 9, the model of the test section had a value of the heat transfer coefficient assigned to the vertical end section and the uninsulated section of the underside of the surface. The values were calculated for a 25 watt power input in run one. The same values were used for the 5 watt input.

A. VERTICAL SECTION

The surface temperature at the vertical section was assumed uniform and equal to 159.4°F. The fluid temperature was assumed to be 159°F. The fluid properties were evaluated at 159°F since the temperature differences were so small. The properties used were taken from Kreith [13]. Pr = 2.5, k = 0.388 Btu/Hr-Ft-°F, $\frac{g\beta}{\nu^2} = 5.2 \times 10^8$ 1/Ft³-°F. The characteristic dimension was chosen as 0.37 inches.

$$Gr_d = \frac{g\beta}{\nu^2}(T_s - T_p) \times \left(\frac{0.37}{12}\right)^3 = 6.07 \times 10^3$$

and

$$Ra_d = Gr_d \times Pr = 6.07 \times 10^3 \times 2.55 = 1.52 \times 10^4.$$

Using correlations given for a vertical surface in McAdams [14]

$$Nu_d = 6.3 = \frac{h\ d}{k}$$

thus

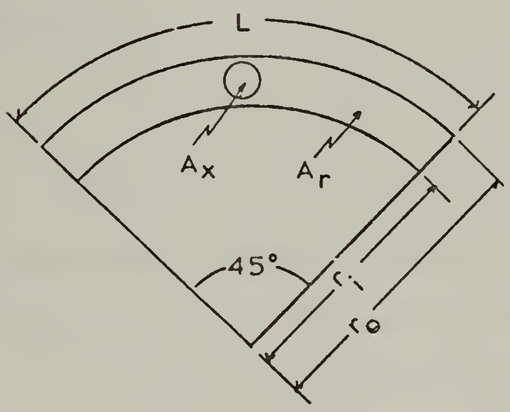
$$h = \frac{Nu_d k}{d} = 79.8 \text{ Btu/Hr-Ft}^2\text{-°F}$$

or

$$h = 80 \text{ Btu/Hr-Ft}^2\text{-}^\circ\text{F}$$

B. BASE SECTION

An equivalent heat transfer coefficient, h_{eqv}, was computed for the base section to account for any fin effect caused by the stud.



$r_i = 1.2$ inches

$r_o = 1.5$ inches

$L = 1.178$ inches

$A_e = A_x + A_r$

$Q_{out} = Q_{fin} + Q_{b.s.}$

where Q_{fin} is the heat loss through the stud and $Q_{b.s.}$ is the heat loss from the base.

$$Q_{fin} = (h_{fin} P k A_x)^{1/2} (T_{fin} - T_{fluid})$$

$$Q_{b.s.} = h_r A_r (T_{b.s.} - T_{fluid})$$

Therefore

$$Q_{out} = (h_{fin} P k A_x)^{1/2} (T_{fin} - T_{fluid}) + h_r A_r (T_{b.s.} - T_{fluid})$$

$$= h_{eqv} A_e (T_{eff} - T_{fluid}).$$

If it is assumed that the temperature on all surfaces is constant and equal the relation reduces to

$$h_{eqv} = \frac{\left(hPkA_x \right)^{1/2}}{A_e} + h \frac{A_r}{A_e}$$

where

$$P = \pi x \frac{3}{16 \times 12} = 0.049 \text{ ft.}$$

$$A_x = \pi/4 \times \left(\frac{3}{16 \times 12}\right)^2 = 0.0002 \text{ ft}^2$$

and

$$A_e = A_x + A_r = \pi(r_o^2 - r_i^2) = 0.002209 \text{ ft}^2$$

therefore

$$A_r = A_e - A_x = 0.002009 \text{ ft}^2$$

Assume that fluid properties are the same as those found for vertical surfaces and that $h_{fin} = 80$ Btu/Hr-Ft^2-°F. Considering a horizontal plate facing downward

$$Nu_L = 0.27(G_r \times P_r)^{1/4} \qquad \text{Ref. [14].}$$

Therefore

$$h_r = 0.27\left[(0.4 \times 5.2 \times 10^8 \times \frac{1.178}{12})\right]^{1/4} \times \frac{0.388 \times 12}{1.178}$$

$$= 28.26 \text{ Btu/Hr-Ft}^2\text{-°F}$$

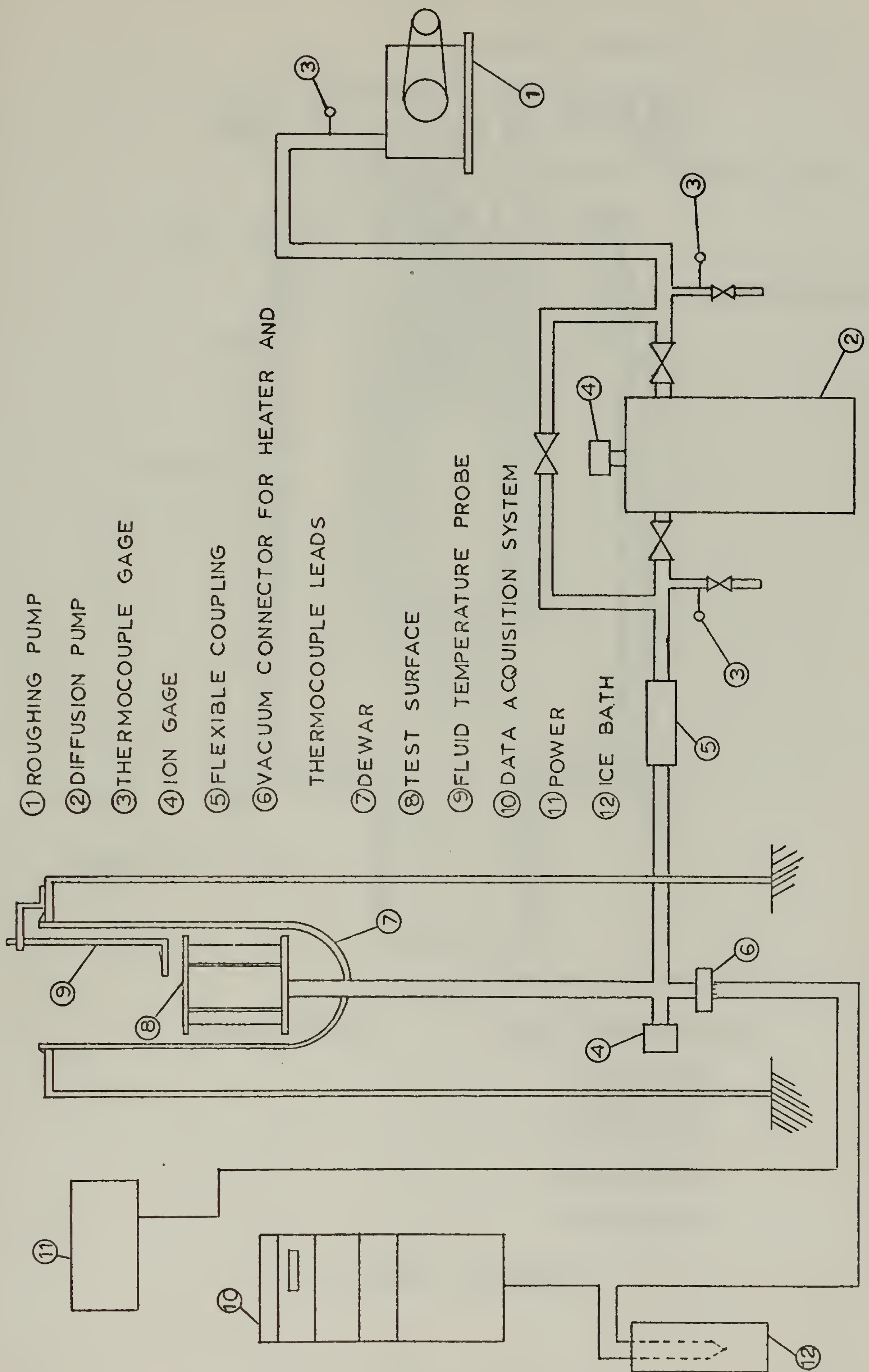
and

$$h_{eqv} = \frac{(80 \times 9.3 \times 0.049 \times 0.002)^{1/2}}{0.002209} + 28.26$$

$$= 52.0 \text{ Btu/Hr-Ft}^2\text{-°F}$$

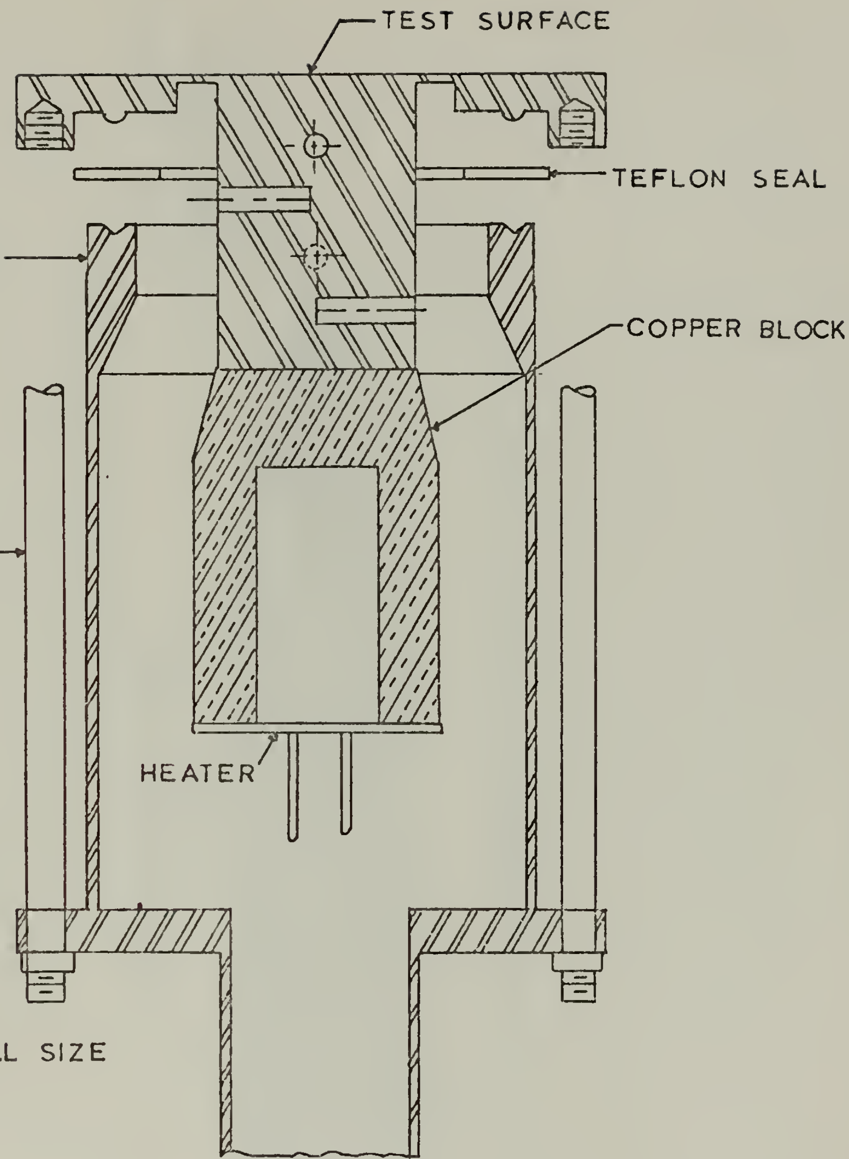
h_{eqv} chosen equal to 52.0 Btu/Hr-Ft^2-°F.

Although the choice of the values of the heat transfer coefficient was based on a number of assumptions, runs made assuming all surfaces insulated resulted in a negligible change.



①ROUGHING PUMP
②DIFFUSION PUMP
③THERMOCOUPLE GAGE
④ION GAGE
⑤FLEXIBLE COUPLING
⑥VACUUM CONNECTOR FOR HEATER AND
THERMOCOUPLE LEADS
⑦DEWAR
⑧TEST SURFACE
⑨FLUID TEMPERATURE PROBE
⑩DATA ACQUISITION SYSTEM
⑪POWER
⑫ICE BATH

FIGURE 1



SCALE: FULL SIZE

THERMOCOUPLE	DISTANCE FROM CENTER LINE TO TEST SURFACE (IN.)
1	0.352 ± 0.001
2	0.600 ± 0.001
3	0.894 ± 0.001
4	1.203 ± 0.001

TEST SECTION ASSEMBLY

FIGURE 2

THERMOCOUPLE PLUG ASSEMBLY

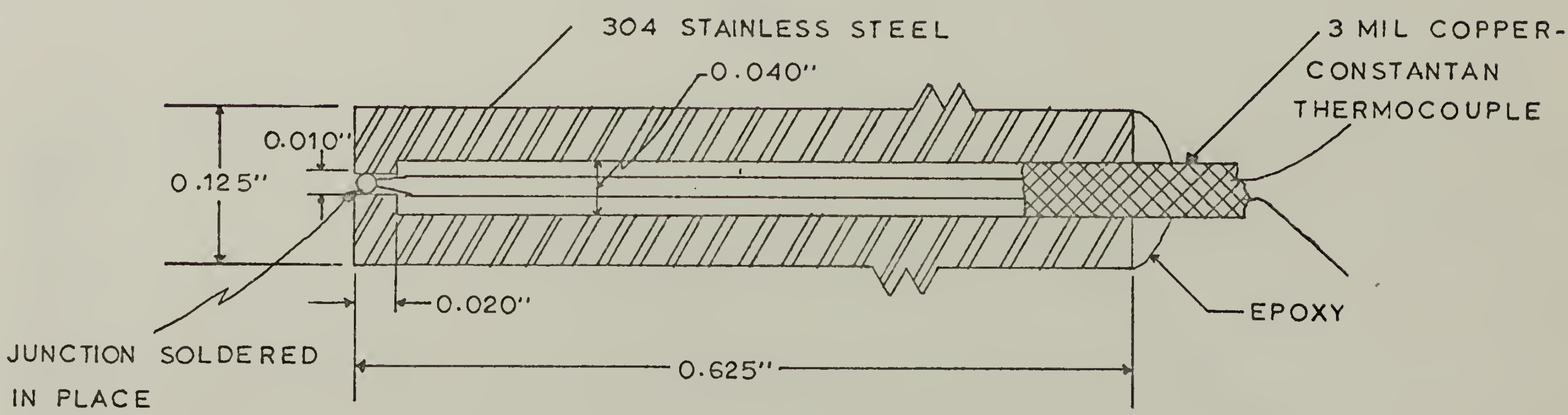


FIGURE 3

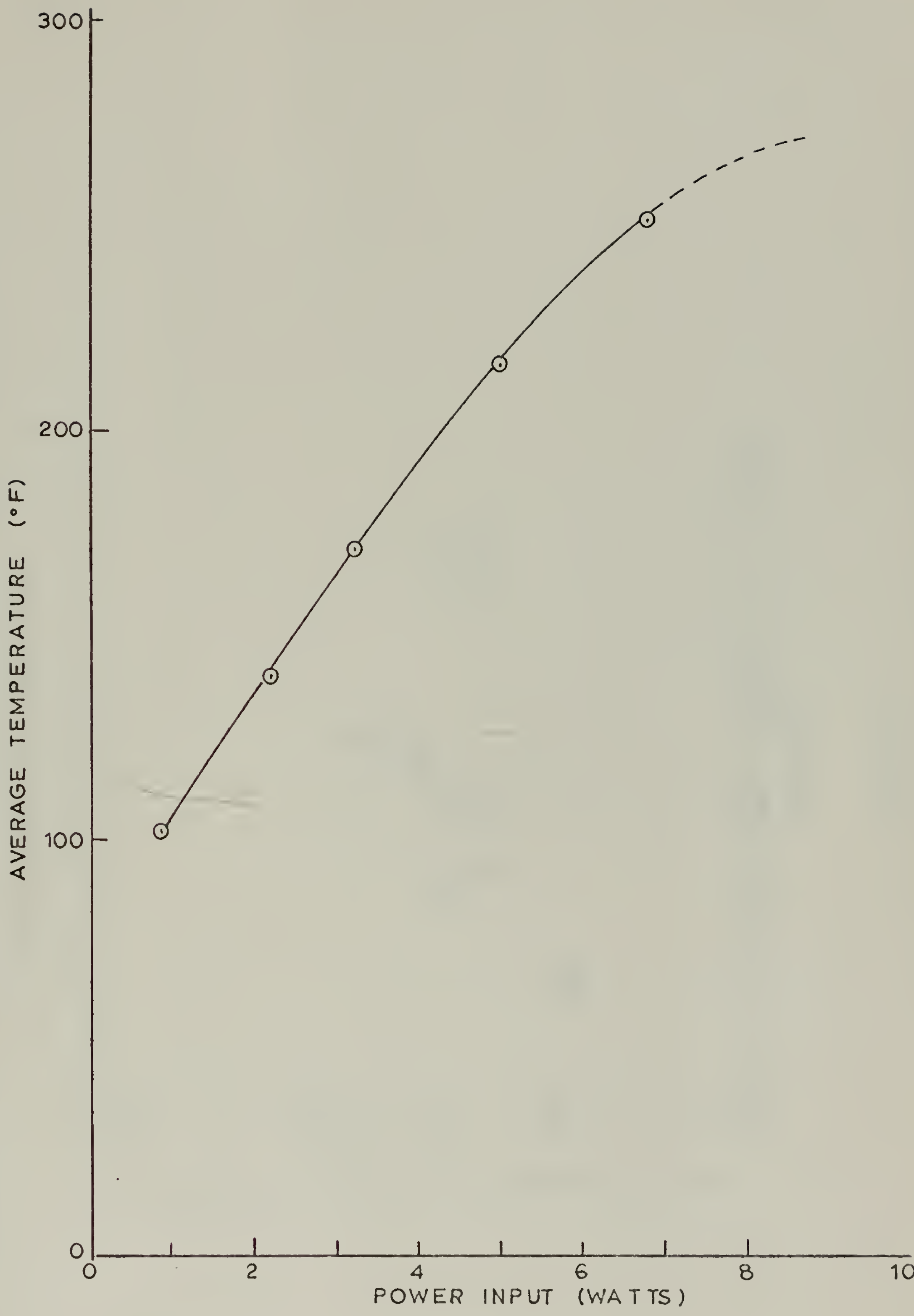

300
200
100
0
AVERAGE TEMPERATURE (°F)
0
2
4
6
8
10
POWER INPUT (WATTS)

FIGURE 4

REPRODUCIBILITY COMPARISON

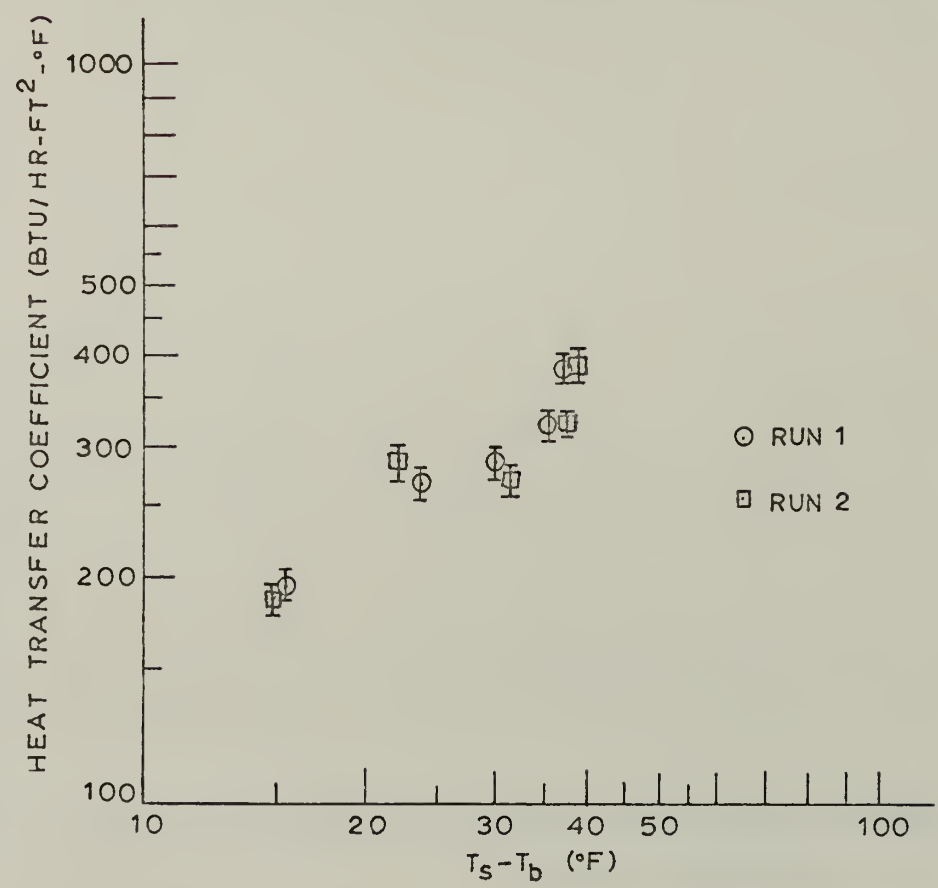


FIGURE 5

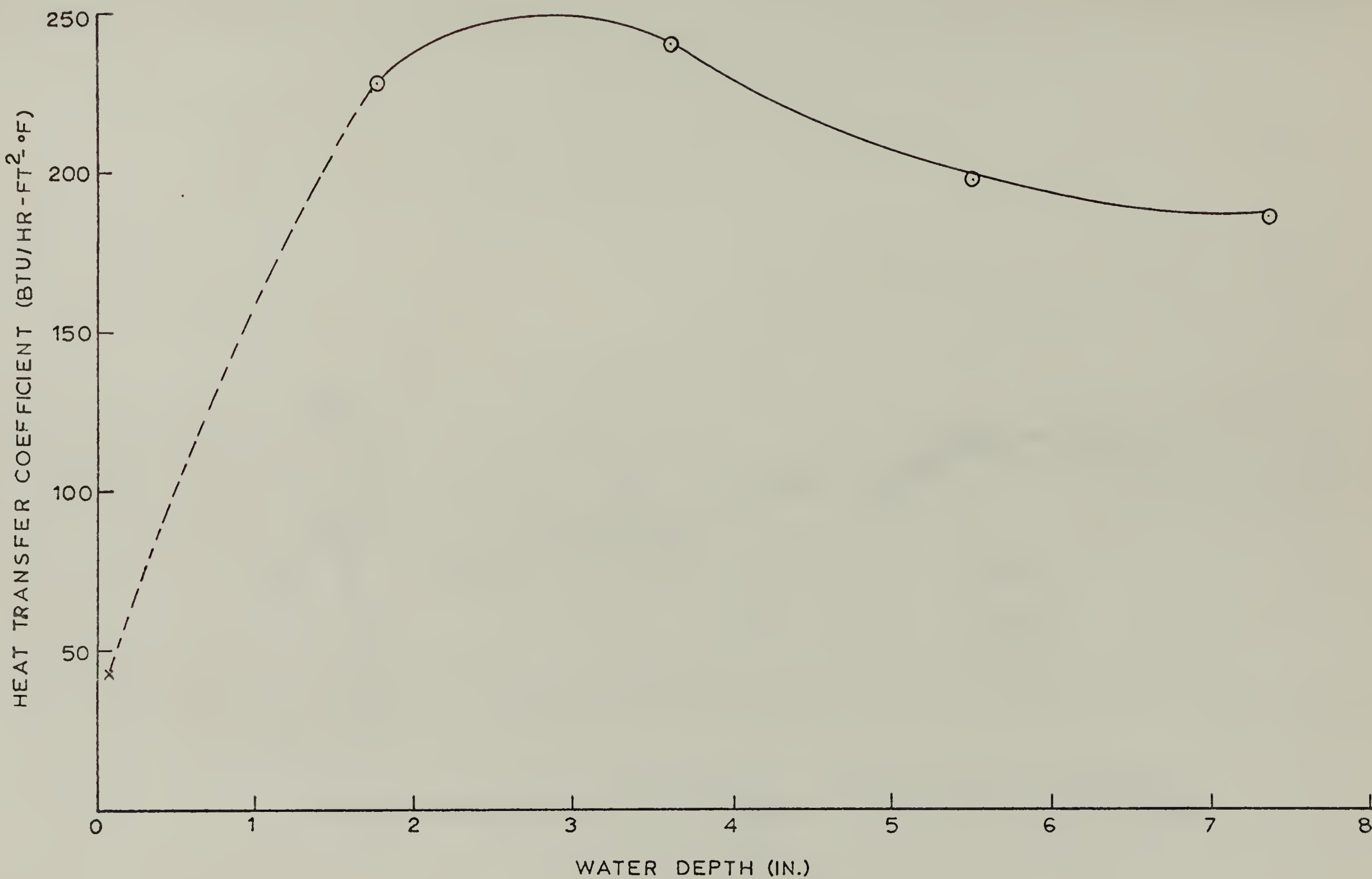

250
200
150
100
50
HEAT TRANSFER COEFFICIENT (BTU/HR-FT2-°F)
0
1
2
3
4
5
6
7
8
WATER DEPTH (IN.)

FIGURE 6

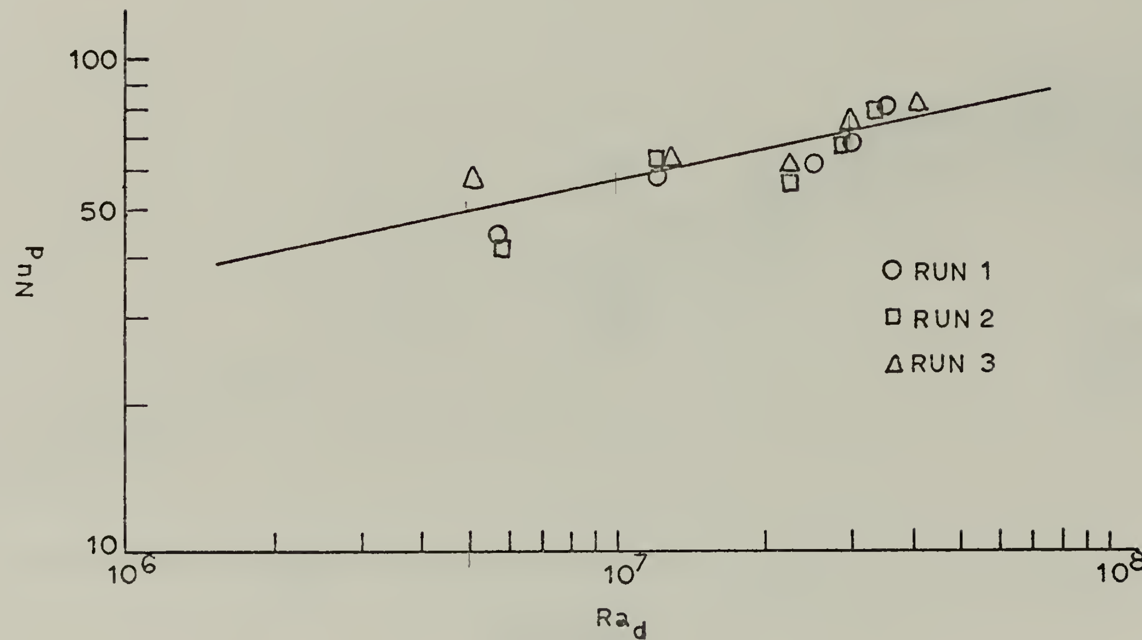
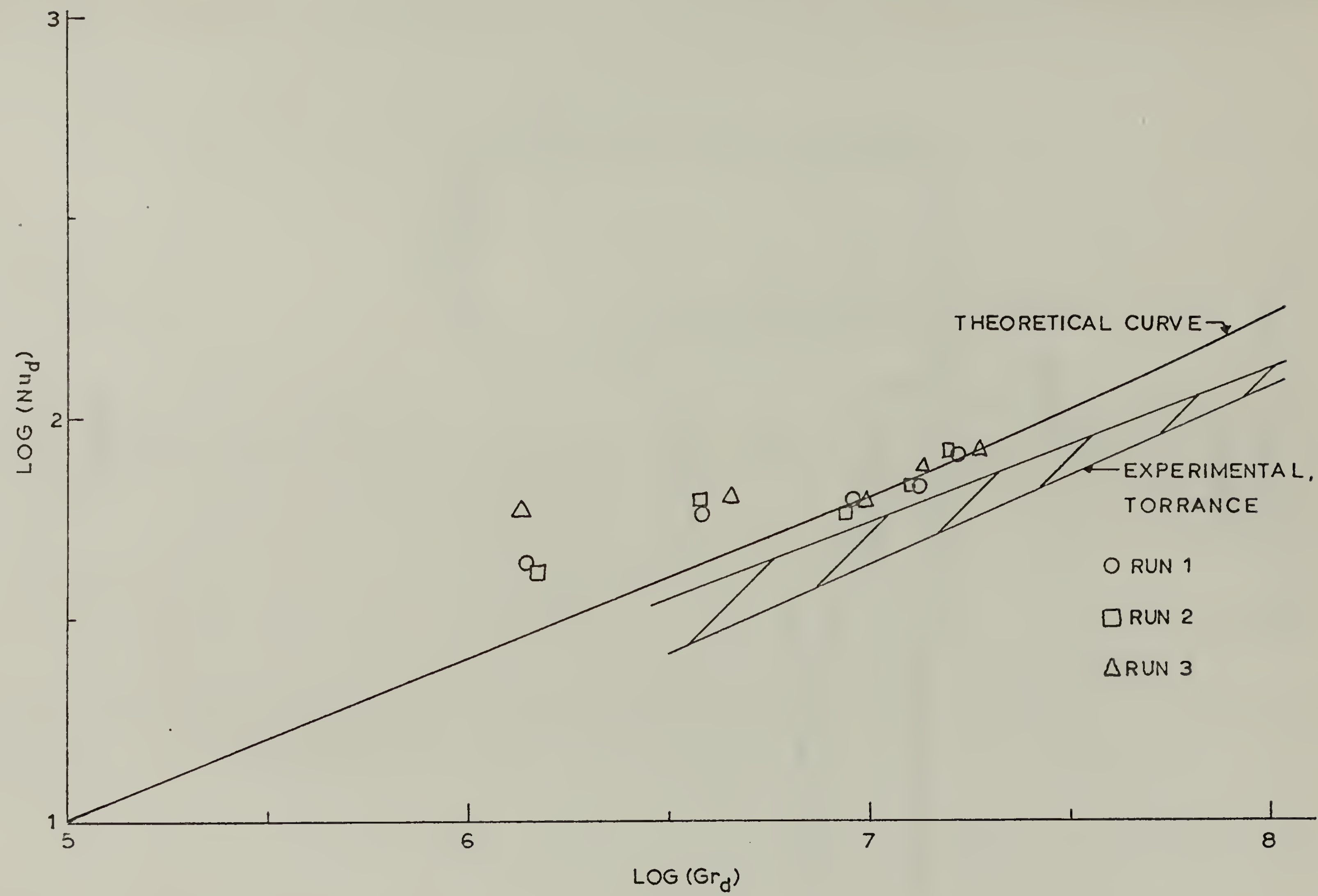


FIGURE 7



THEORETICAL CURVE
EXPERIMENTAL,
TORRANCE
O RUN 1
□ RUN 2
Δ RUN 3
LOG (Nu_d)
LOG (Gr_d)
3
2
1
5
6
7
8

FIGURE 8

COMPUTER MODEL OF TEST SURFACE

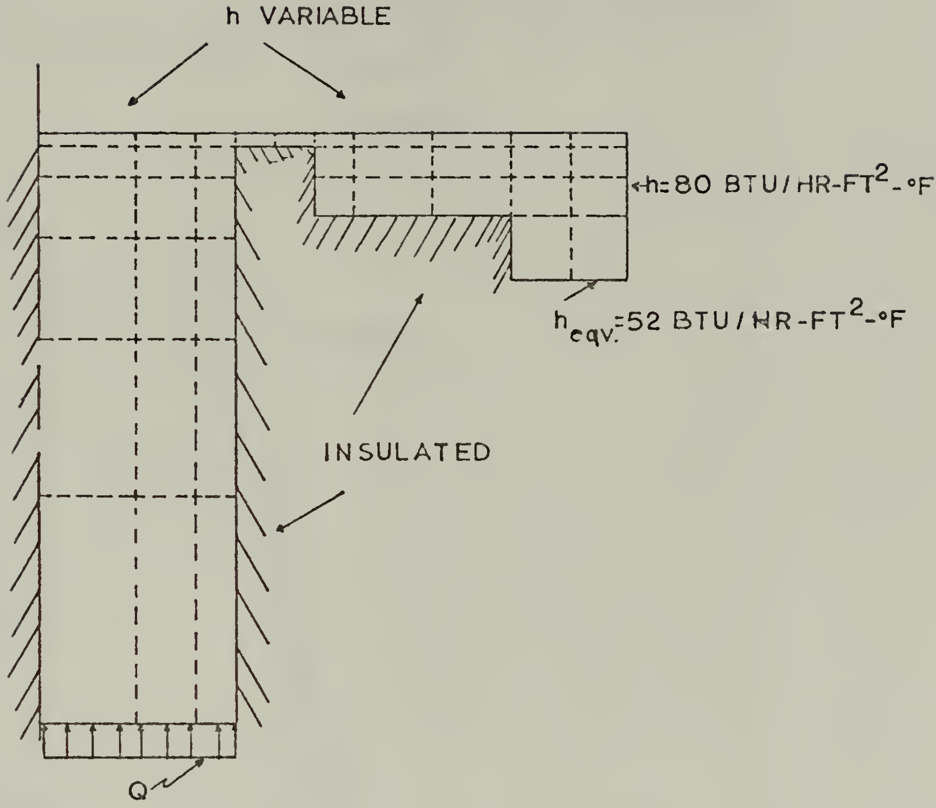


FIGURE 9

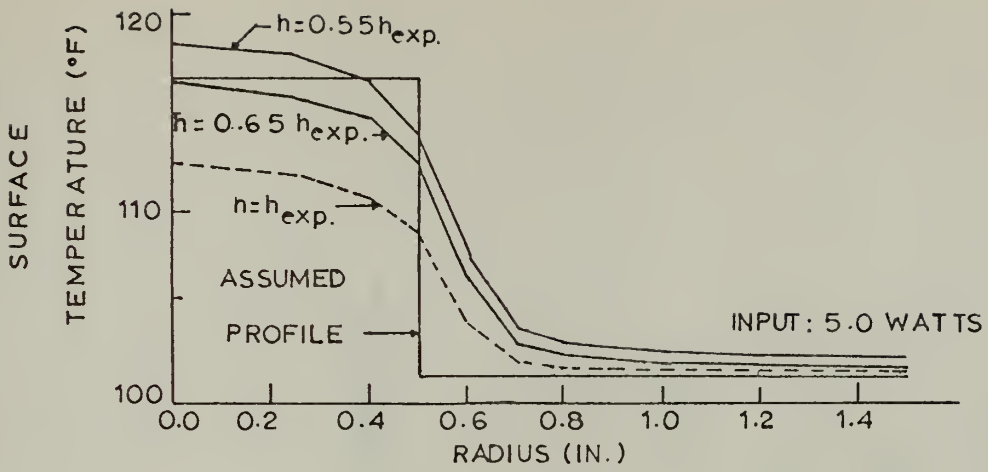


FIGURE 10

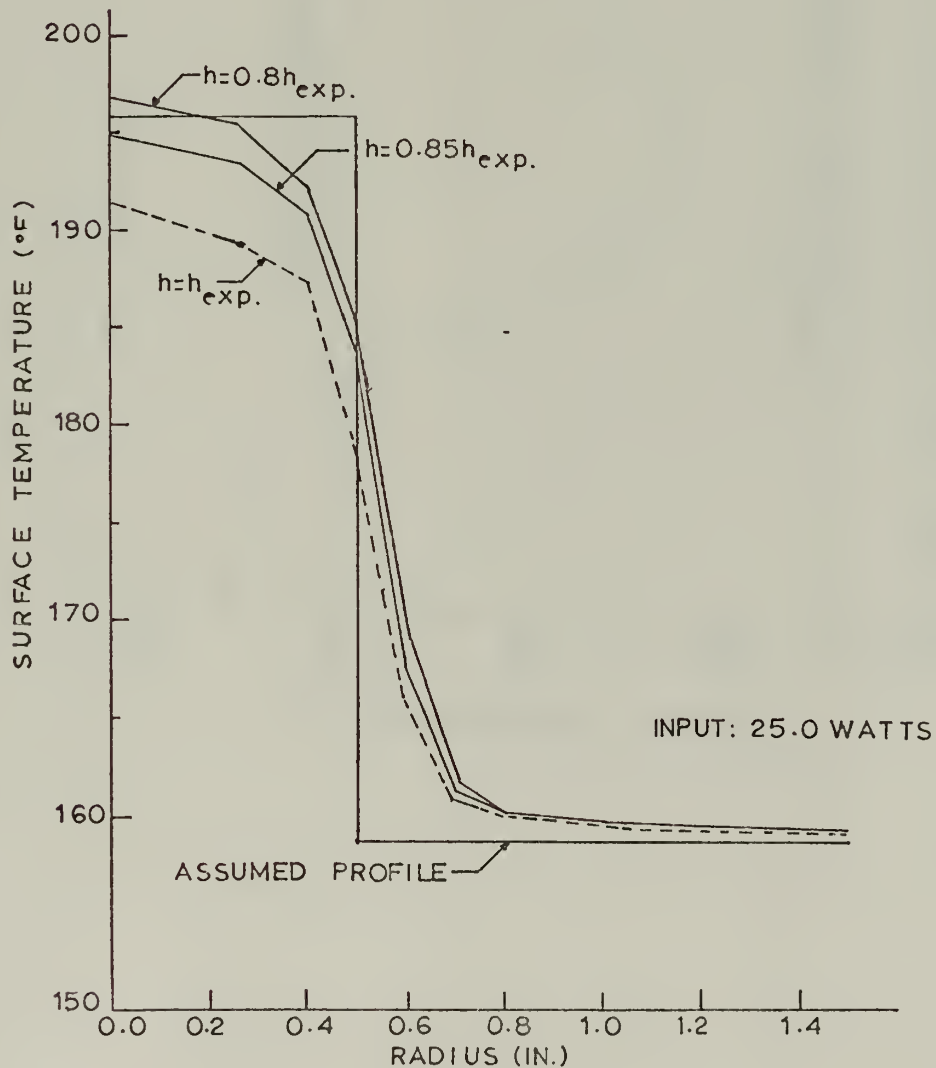


FIGURE 11

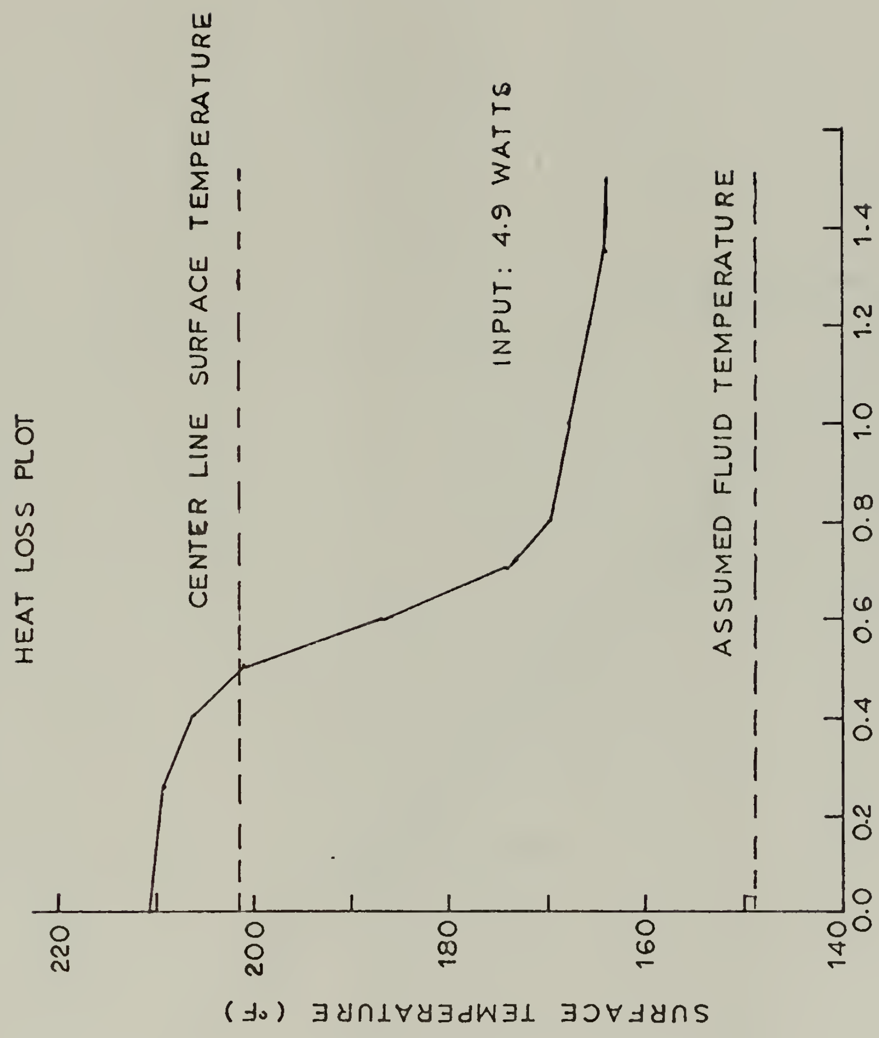
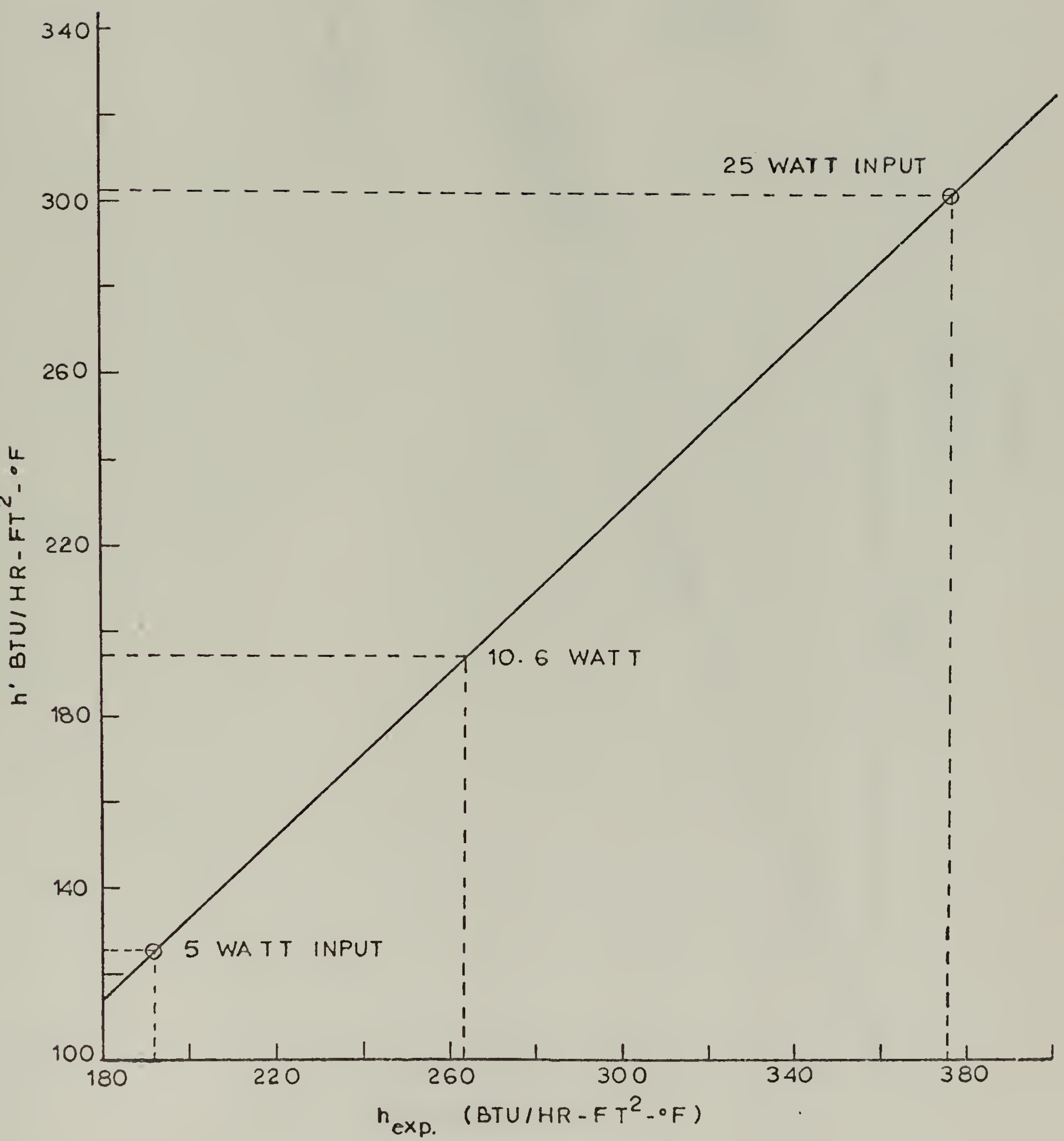
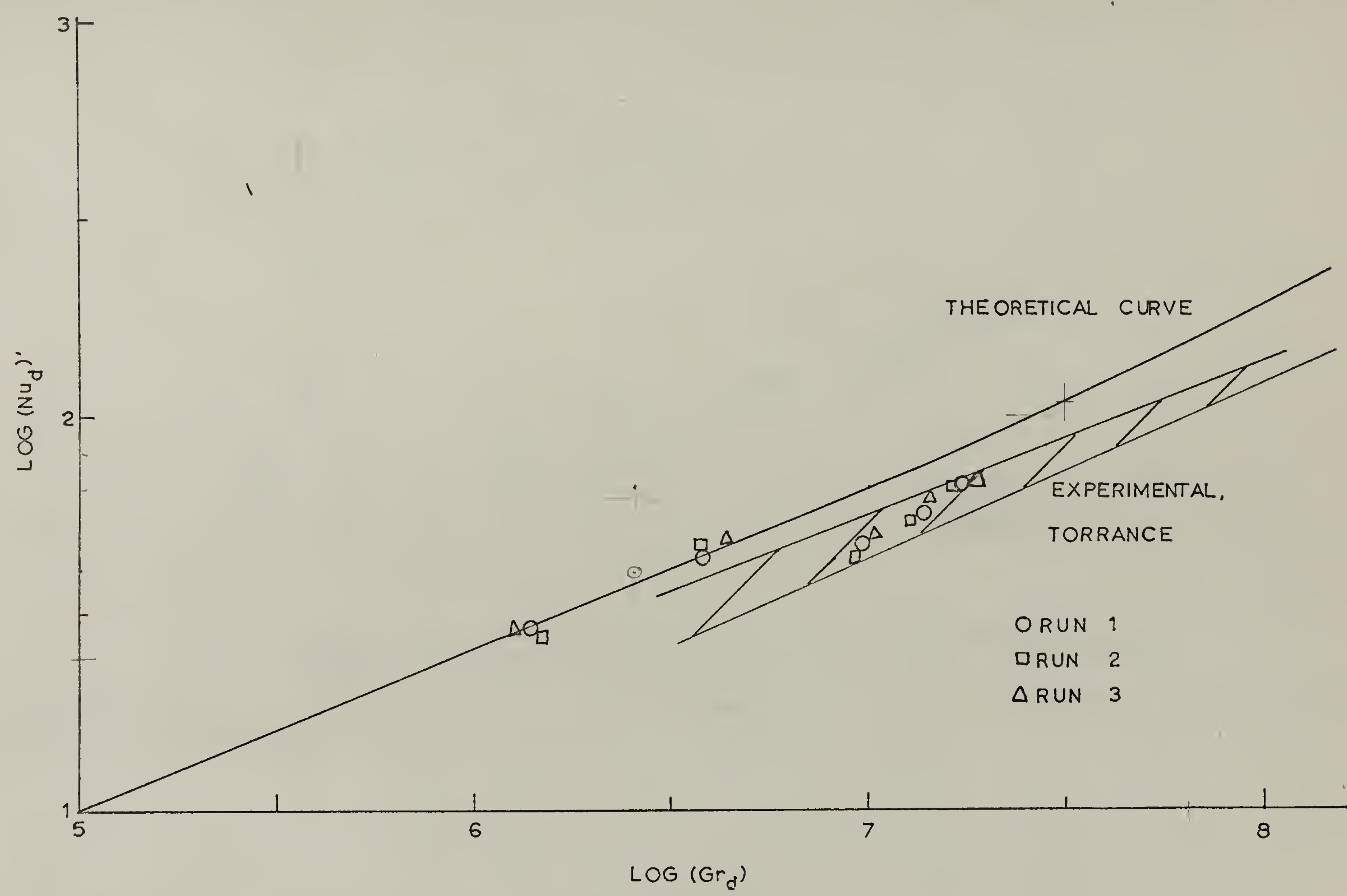

HEAT LOSS PLOT
CENTER LINE SURFACE TEMPERATURE
INPUT: 4.9 WATTS
ASSUMED FLUID TEMPERATURE
SURFACE TEMPERATURE (°F)
220
200
180
160
140
0.0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
RADIUS (IN.)

FIGURE 12



340
300
260
220
180
140
100
180
220
260
300
340
380
25 WATT INPUT
10. 6 WATT
5 WATT INPUT
h' BTU/HR-FT^2-°F
$h_{exp.}$ (BTU/HR-FT^2-°F)

FIGURE 13



THEORETICAL CURVE
EXPERIMENTAL, TORRANCE
O RUN 1
□ RUN 2
Δ RUN 3
LOG $(Nu_d)'$
LOG (Gr_d)
1
2
3
5
6
7
8

FIGURE 14

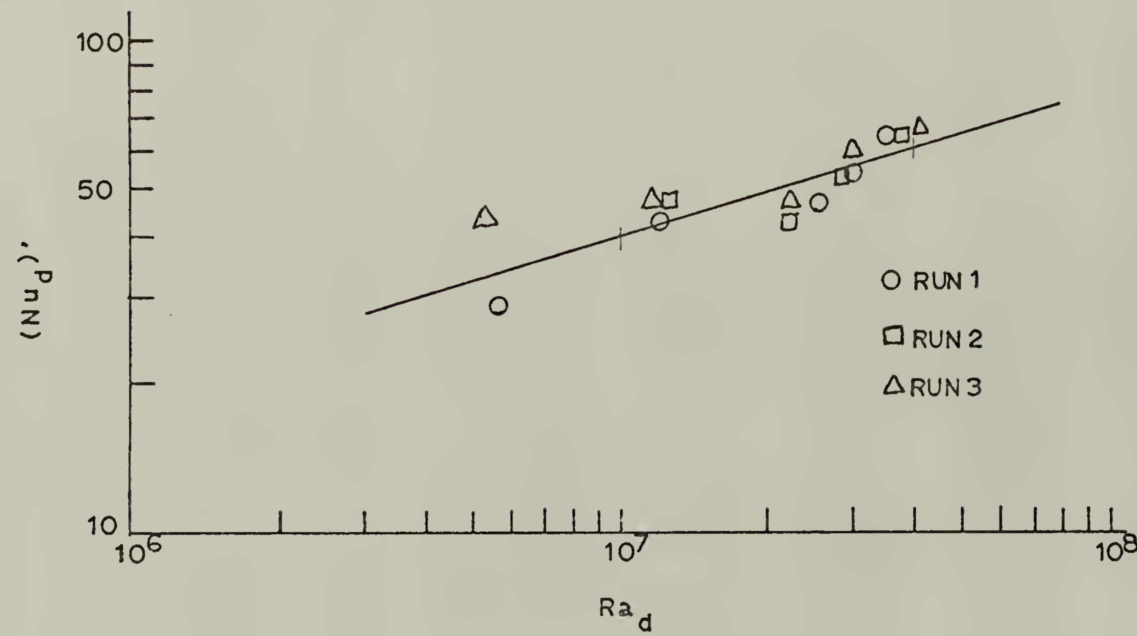


FIGURE 15

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Security Classification

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1. ORIGINATING ACTIVITY (Corporate author)

Naval Postgraduate School
Monterey, California 93940

2a. REPORT SECURITY CLASSIFICATION

Unclassified

2b. GROUP

3. REPORT TITLE

Natural Convection Heat Transfer from a Horizontal Disk in a Cylindrical Enclosure

4. DESCRIPTIVE NOTES (Type of report and, inclusive dates)

Master's Thesis, December 1971

5. AUTHOR(S) (First name, middle initial, last name)

Duane S. Ducan, Jr.

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
December 1971	58	14

8a. CONTRACT OR GRANT NO.

b. PROJECT NO.

c.

d.

9a. ORIGINATOR'S REPORT NUMBER(S)

9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)

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11. SUPPLEMENTARY NOTES

12. SPONSORING MILITARY ACTIVITY

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13. ABSTRACT

This thesis studied natural convection in water from a horizontal mirror finished disk, heated in the center and surrounded by a cylindrical enclosure. The test section was designed to insure a one dimensional heat flux. Measurements were taken with power inputs varying from 25 to 5 watts.

Fluid depth had a definite effect on the heat transfer. An apparent maximum value existed at a ratio of enclosure radius to fluid depth of one. Comparison of the data with existing correlations however, was poor which led to the questioning of the validity of the assummed one dimensional heat flux. Use of a finite element computer program demonstrated that two dimensional effects were important. Subsequent modification of the heat transfer coefficient to account for this variation gave a correlation more in agreement with those existing in the literature.

DD FORM 1 NOV 65 1473 (PAGE 1)

S/N 0101-807-6811

Security Classification

A-31408

KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Natural Convection						
Horizontal Heat Transfer						

Thesis 133808
D7889 Duncan
c.1 Natural convection
heat transfer from a
horizontal disk in a
cylindrical enclosure.